The influence of airflow on the effectiveness of rotary heat exchangers operating under high-speed rotor conditions

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Abstract. The paper presents an analysis of heat and mass transfer processes occurring inside the rotary heat exchanger operating under high-speed rotor conditions for different values of the airflow rate. For this purpose the original mathematical α -model was used. Conducted computer simulations allowed to determine the influence of Number of Transfer Units (NTU) of airflow on the temperature effectiveness as well as on the distribution of different active heat and mass transfer zones: “dry”, “wet” and “frost”. It was found that the increase of the values of NTU strictly affects the increase of the effectiveness of heat recovery. Another issue emerging from this study is the fact that in the certain range of low values of NTU there is no “dry” area created. It was established that at low values of NTU ($NTU \approx 1$) “frost” area extremum and sharp drop in the “frost” area accumulation are observed.

1 Introduction

Rotary heat exchangers (Fig. 1) belong to the group of commonly used devices for heat recovery in ventilation and air conditioning systems. The main element of their construction is a rotor consisting of wound aluminium foil with high thermal conductivity. The thickness of the aluminium foil is about 0.1mm. The height of the fins through which the air flows depends on the level of air pollution and ranges from 1.5 to 2.5 mm.

The general operation principle of such kind heat exchangers is based on the initial transfer from the warmer air stream to the matrix located on the return air side (4 in Fig. 1). The heated part of the matrix is then transferred by rotation to the supply air side (3 in Fig. 1), where it releases the heat to the cooler air stream. The rotation of the rotor is caused by the operation of an electric motor installed with the heat exchanger in the housing of heat recovery section. Heat recovery efficiency of the rotary heat exchangers can be up to 80% or even more. For this reason, many researchers devoted a lot of attention to the analysis of heat wheels operation.

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Fig. 1. The construction of a rotary heat exchanger: 1 – metal housing, 2 – rotor, 3 – supply air side, 4 – return air side, 5 – drive belt, 6 – electric motor.

Fathieh et al. [1] performed the cyclic and single step change transient experiments for predicting the sensible effectiveness of air-to-air heat wheel. Abe et al. [2] presented the analytical model for predicting the rotary heat exchanger effectiveness. For this purpose authors used only the characteristics measured on the same non-rotating wheel exposed to a gradual change in temperature and relative humidity parameters. In the next paper written by Abe et al. [3] the authors verified the accuracy of the proposed transient test method for air-to-air energy wheels. Simonson et al. [4, 5] presented equations for determining sensible, latent and total effectiveness of energy wheel. The authors claimed that the presented correlations are characterized by compliance at the level of 2.5%. Seo et al. [6] presented a simple model for estimating the rotary heat exchanger effectiveness. They concluded that the maximum calculation error does not exceed 5% and is compared with a widely accepted reference in the literature.

Thermal effectiveness of the rotary heat exchanger is not constant and depends on many factors. In addition to the geometrical dimensions, the properly adjusted speed of rotation and the structure of the matrix material, special attention should be paid to the temperature and humidity parameters of both air streams. These two factors play a key role in determining the nature of thermodynamic processes occurring inside the heat exchanger, especially under frosting conditions. It should be noted, that the subject of freezing of heat exchangers was discussed in many different papers.

Rafati Nasr et al. [7] provided an overview of frost formation and frost prevention techniques in air-to-air heat and energy exchangers. In the paper written by Liu et al. [8] the simplified theoretical model for predicting the inlet conditions under which frost will form in the cross-flow plate heat and energy exchangers was presented. Anisimov et al. [9] developed the original mathematical model used for the simulation of cross-flow heat exchanger performance under frost formation conditions. Jedlikowski et al. [10] presented the original model for predicting heat and mass transfer in the channels of counter-flow plate heat exchanger under sub-zero outdoor air temperature conditions. Moreover, in the next paper Jedlikowski and Anisimov [11] analysed three frost control techniques for cross-flow heat exchangers. In further paper written by Pacak et al. [12] the analysis of coupled heat and mass transfer inside the counter-flow heat exchanger under low outdoor air temperatures conditions was presented. The authors analysed and compared two different

frost prevention strategies. Bilodeau et al. [13] presented the mathematical model for predicting of rotary heat exchanger performance. They concluded that the absolute humidity is the prevailing parameter to characterize the frosting phenomenon. Tun [14] analysed the rotary heat exchanger under “dry” heat transfer and condensation conditions (in the form of water film and frost layer). The author concluded that the threshold outdoor air temperature level under which frost accumulation occurs is significantly lower in the case of heat and energy wheel.

Despite the existence of many studies describing the operation of rotary heat exchangers, so far there is no detailed analysis of the nature of the thermodynamic processes occurring inside the matrix of heat exchanger in relation to the airflow rate.

2 Analysis of creating different active heat and mass transfer zones

In order to analyse rotary wheel performance it is necessary to establish year-round operating conditions under which different combinations of active heat and mass transfer zones “dry”, “wet” and “frost”) arise (see Fig. 2).

Fig. 2. Schematic diagram of the rotary heat exchanger.

The presence of these areas depends mainly on the surface temperature of the heat exchanger matrix in relation to the two characteristic values of dew point and freezing point temperatures. In simple terms, it can be stated that if the local temperature of the matrix surface of the heat exchanger is higher than the dew point temperature $t_j^{DP} < t_{pj}$, a “dry” heat exchange takes place and only a “dry” zone is created in the heat exchanger matrix. On the other hand, if there is a reverse relation $t_j^{DP} > t_{pj}$, water-vapour condensation takes place on the solid surface on the supply/exhaust air side in the form of the water film under conditions $(0^\circ\text{C} < t_{pj} < t_j^{DP})$ or in the form of the frost layer (water-vapour sublimation) under conditions $((t_{pj} < t_j^{DP}) \cap (t_{pj} < 0^\circ\text{C}))$.

One of the most complex combinations is the ability to create three zones at the same time [9–12]. However, the relative size of these zones may vary depending on the airflow rate due to the significant influence of the contact time of the air streams with the matrix.

For this reason, the authors of the study decided to analyse the variability of the active heat and mass transfer areas and determine the effectiveness of the device depending on the air flow rates. One of the perspective way of carrying out such an analysis is to maintain a constant rotor speed while increasing the airflow. For this purpose, an original mathematical model of a rotary heat exchanger operating under high-speed rotor conditions was developed. The model is based on the original model of the counter-flow heat exchanger [10] and assumes non-oscillating mode of operation. A detailed description of the model was presented in our previous paper [15]. On this basis, a computer program was written to simulate the operation of the rotary heat exchanger.

3 Results of simulation

Numerical simulations were performed by entering the following input data into the calculation program (Table 1).

Table 1. The input geometrical and thermodynamic parameters.

Rotor size parameters			Airflow parameters	
height of housing	mm	800	outdoor air	
width of housing	mm	900	temperature	$t_{li} = -20^{\circ}\text{C}$
wheel diameter	mm	715	relative humidity	$RH_{li} = 100\%$
wheel depth	mm	300	return air	
thickness of material	mm	0.10	temperature	$t_{2i} = +20^{\circ}\text{C}$
height of waves	mm	2.00	relative humidity	$RH_{2i} = 40\%$

The airflow rate has been converted to the Number of Transfer Units, commonly used to analyse the heat exchangers operation [16]. For this purpose, the following mathematical transformation were used (1)–(4).

Number of Transfer Units can be determined by the following equation:

$$NTU = \alpha F / W \tag{1}$$

Heat capacity rate can be calculated as follows:

$$W = Gc_p \tag{2}$$

Moist air mass flow rate can be defined as follows:

$$G = \rho \dot{V} \tag{3}$$

Volumetric flow rate for one airflow side can be calculated as follows:

$$\dot{V} = wA_{eff} / 2 \tag{4}$$

The entered data and the above-mentioned formulas made it possible to perform a number of numerical simulations for NTU in the range of 0...20 (Figs. 3–5).

Fig. 3. Percentage distribution of different areas inside the rotary heat exchanger expressed as a function of Number of Transfer Units of supply/return air (a) “dry” area (b) “wet” area.

Fig. 4. Percentage distribution of different areas inside the rotary heat exchanger expressed as a function of Number of Transfer Units of supply/return air (a) “frost” area (b) “frost” area accumulation.

Fig. 5. Graphical representation of (a) the local minimum and maximum temperatures of plate surface (b) the temperature effectiveness of the rotary heat exchanger, expressed as a function of Number of Transfer Units of the supply/return airflow

On the basis of the graphs (Figs. 3–4) it was found that in the range of low values $NTU=0...1$, only condensation zones (“wet” and “frost”) are observed. For NTU values close to zero, the volume flow rate of two airflows reaches such high values that effective contact of these airflows with the plate surface is practically impossible (Fig. 5(a)). The

inlet dew point temperature for the assumed return airflow parameters (Table 1) is equal $t_2^{DP} = 6^\circ\text{C}$. The local plate surface temperature of the rotary heat exchanger matrix in each analysed cross section will reach lower values ($t_{pj} = 0.3...0.7^\circ\text{C}$) (Fig. 5(a)) but never exceed the freezing point temperature ($0^\circ\text{C} < t_{pj} < t_j^{DP}$). In this case, only a “wet” zone will be created inside the heat exchanger matrix (Fig. 2). Although the heat exchanger matrix is completely covered with water film, the nature of heat and mass transfer processes is very limited. Due to the lack of the water vapour accumulation on the matrix surface the local latent heat released on the return air side, despite reaching very high values, is entirely directed to the airflow on the supply air side during evaporation process. In this regard, the latent heat transfer does not impact on the temperature effectiveness of heat recovery which is very low under such inlet airflow conditions ($\varepsilon = 0.03$, see Fig. 5(b)). Analysing the further course of the developed curves, in the direction to the right of zero value, the initially dominant “wet” zone begins gradually to decrease at the expense of creating an additional “frost” zone (Fig. 3(b) and Fig. 4(a)). Due to the slightly longer contact time of the air streams with the rotary heat exchanger matrix, the local plate surface temperatures, observed at the outlet areas on the return air side, reaches values lower than freezing point ($t_{pj} < 0^\circ\text{C} < t_j^{DP}$) (see Fig. 5(a)). As a result, the “frost” area begins to rise rapidly, reaching about 50% of the total surface area of the matrix (Fig. 4(a)). This phenomenon is also accompanied by the frost accumulation on the matrix surface, indicating the size of the “frost” area, in which frost layer thickness is rising despite the partial frost sublimation on the supply air side (Fig. 4(b)). The lower the value of the frost layer thickness and “frost” accumulation surface, the higher the ability to remove frost on supply air side. In this case, only two regions (“wet” and “frost”) will be formed inside the heat exchanger matrix. Due to increase of the NTU values, the effectiveness of the heat exchanger will also gradually increase (up to 0.30) (Fig. 5(b)). Comparing two graphs of the percentage frost distribution (Figs. 4(a) and 4(b)), it can be seen that near the value $\text{NTU}_j \approx 1$ a “frost” area extremum is observed (Fig. 4(a)) and sharp drop in the “frost” accumulation area (Fig. 4(b)). It should be also noted, that a “dry” area begins to form at the expense of reducing the “wet” and “frost” zones (Fig. 3(a)). This is due to the fact, that in a certain cross sections of the rotary heat exchanger matrix (near inlet return airflow area), the local temperature of the matrix surface is higher than the dew point temperature ($t_{pj} > t_j^{DP}$) (Fig. 5(a)). The presence of the “dry” zone is directly connected with the phenomenon of gradual removal of the frost layer by its sublimation. However, only partial elimination of the “frost layer” is possible at this value, confirmed by an almost double reduction of the “frost” area accumulation (Fig. 4(b)). For the values of $\text{NTU}_j = (1...4)$, a significant increase of the “dry” area combined with a decrease of the “wet” area and a gradual decrease of the “frost” area was established. Moreover, a significant increase in the “frost” accumulation area was also observed. The heat exchanger under these operating conditions achieved temperature effectiveness in the range of $\varepsilon = (0.35...0.70)$. The values of $\text{NTU}_j > 4$ are accompanied by a gradual increase of the “dry” area with a simultaneous decrease of the “frost” area and its accumulation, as well as a decrease in the relatively low share of “wet” area. In addition, due to a further increase in the NTU values of return air, it was possible to achieve a very high effectiveness of the rotary heat exchanger $\varepsilon = 0.90$ (Fig. 5(b)).

4 Conclusions

The proper exploitation of the HVAC systems with the heat recovery devices should be based on the high effectiveness while ensuring safe operation conditions. For this purpose the detailed analyses of the thermodynamic processes occurring inside the heat recovery exchangers were conducted. The paper is concerned with the analysis of heat and mass transfer processes and the temperature effectiveness of the rotary heat exchanger under high rotational speed conditions for constant inlet airflow parameters.

On the basis of the original mathematical α -model the series of numerical simulations was accomplished. The results allow to draw a few insights:

- The nature of heat and mass transfer processes inside the rotary heat exchanger depending on the number of transfer units of the supply/return airflow were presented.
- It was shown that the appropriate contact time of air streams with the matrix of the heat exchanger has a significant impact on the percentage distribution of active heat and mass transfer areas and the effectiveness of heat recovery device.
- For $NTU_j \approx 1$ the presence of the largest area covered with “*frost*” and a steep drop in the maximum size of “*frost*” area accumulation at the total absence of “*dry*” area were found. In this case the maximum value of the matrix surface temperature is equal to the dew point temperature ($t_{pj}^{\max} = t_j^{DP} = 6^\circ\text{C}$).
- In the $NTU_j = (1...4)$ range, the dynamic growth of “*dry*” area and “*frost*” area accumulation was revealed, associated with a dynamic decrease of the “*wet*” area. Moreover in further parts of the charts ($NTU_j > 4$) (Figs. 3, 4) the course of active heat and mass transfer areas is characterized by gradual stabilization.
- The results obtained will be used for further optimization studies and they will allow to evaluate the possibility of using a rotary heat exchanger depending on different climatic conditions.

Nomenclature

A_{eff} – effective area of wheel diameter (airflow surface), m^2 ; c_p – specific heat capacity, $\text{J}/(\text{kg}\cdot\text{K})$; ε – effectiveness, dimensionless; F – heat or heat and mass transfer surface area, m^2 ; F_o – total surface area, m^2 ; $\bar{F} = (F/F_o) \cdot 100\%$ – heat or heat and mass transfer surface area relates to total surface area, %; G – moist air mass flow rate, kg/s ; t – temperature, $^\circ\text{C}$; L_{X_1} – channel of heat exchanger in X_1 direction, m ; L_{X_2} – channel length of heat exchanger in X_2 direction, m ; $\dot{V}$ – volumetric flow rate, m^3/s ; w – air velocity, m/s ; W – heat capacity rate of fluid, W/K ; X_1 – Cartesian coordinate in X_1 direction (along outdoor airflow direction), m ; $\bar{X}_1 = X_1/L_{X_1}$ – relative X_1 coordinate, dimensionless; X_2 – Cartesian coordinate in opposite to X_1 direction, m (along return airflow direction); $\bar{X}_2 = X_2/L_{X_2}$ – relative X_2 coordinate, dimensionless; α – convective heat transfer coefficient, $\text{W}/(\text{m}^2 \text{K})$; ε – temperature effectiveness of heat exchanger, dimensionless; ρ – density of moist air, kg/m^3

Subscripts: 1 – outdoor airflow; 2 – return airflow; j – number of airflow channel; *eff* – effective; *dry* – dry area; *frost* – frost area, i – inlet; o – outlet, p – matrix surface/under constant pressure conditions; *wet* – wet area.

Superscripts: *acc* – accumulation; *DP* – referenced to dew point temperature; *max* – maximum value; *min* – minimum value.

Acronyms: NTU – Number of Transfer Units; *RH* – Relative Humidity, %.

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References

1. F. Fathieh, R. W. Besant, R. W. Evitts, C. J. Simonson, *Int. J. Heat Mass Transf.* **87**, 312–326 (2015)
2. O. O. Abe, C. J. Simonson, R. W. Besant, W. Shang, *Int. J. Heat Mass Transf.* **49**, 52–62 (2006)
3. O. O. Abe, C. J. Simonson, R. W. Besant, W. Shang, *Int. J. Heat Mass Transf.* **49**, 63–77 (2006)
4. C. J. Simonson, R. W. Besant, *Int. J. Heat Mass Transf.* **42**, 2161–2170 (1999)
5. C. J. Simonson, R. W. Besant, *Int. J. Heat Mass Transf.* **42**, 2171–2185 (1999)
6. J. W. Seo, D. Y. Lee, D. S. Kim, *Int. J. Heat Mass Transf.* **120**, 1358–1364 (2018)
7. M. Rafati Nasr, M. Fauchoux, R. W. Besant, C. J. Simonson, *Renew. Sustain. Energy Rev.* **30**, 538–554 (2014)
8. P. Liu, M. Rafati Nasr, G. Ge, M. Justo Alonso, H.M. Mathisen, F. Fathieh, C. J. Simonson, *Energy Build.* **110**, 404–414 (2016)
9. S. Anisimov, A. Jedlikowski, D. Pandelidis, *Int. J. Heat Mass Transf.* **90**, 201–217 (2015)
10. A. Jedlikowski, S. Anisimov, J. Danielewicz, M. Karpuk, D. Pandelidis, *Int. J. Heat Mass Transf.* **108**, 585–613 (2017)
11. A. Jedlikowski, S. Anisimov, *Appl. Therm. Eng.* **116**, 731–765 (2017)
12. A. Pacak, A. Jedlikowski, M. Karpuk, S. Anisimov, *Int. J. Heat Mass Transf.* **133**, 842–860 (2019)
13. S. Bilodeau, P. Brousseau, M. Lacroix, Y. Mercadier, *Int. J. Heat Mass Transf.* **42**, 2605–2619 (1999)
14. E. Tunå, *Numerical Analysis of Condensation and Frost in Rotary Heat Exchangers* (Master Thesis, Norwegian University of Science and Technology, Department of Energy and Process Engineering, 2014)
15. P. Kanaś, A. Jedlikowski, S. Anisimov, *SN Appl. Sci.* **1**, 526 (2019)
16. W. M. Kays, A. L. London, *Compact Heat Exchangers* (3rd ed., McGraw-Hill, New York, 1984)